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Hydrodynamic Seal on the Basis of a Cylindrical Layer of the Compressible Fluid with a Running Wave

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Abstract

The hydrodynamic seal is based on a cylindrical layer of the compressible fluid with a running wave, which, unlike the conventional hydrodynamic seal, has the characteristics that do not depend on the rotation speed of the sealed shaft. The flow of the compressible fluid in the cylindrical gap is concerned, the pressure distribution in the layer are obtained, pressure drops (counterpressure) along the axis of the shaft depending on the gas content in the liquid are determined.

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1. Introduction.

Contactless liquid seal is the device, the sealing action of which is achieved as a result of energy loss during the motion of the fluid in the channels formed by the seal elements. Most often the non-contact seals are used for sealing rotating shafts. The medium filling the seal, is treated in the apparatus of the working fluid or process fluid, preventing contact of the fluid with the environment. According to the principle of the non-contact seal can be static or dynamic. The work of static compression is connected with the forces arising from the contact of fluid with the sealing surface and overcome local resistances. The work of contactless dynamic seal is associated with the pressure (backpressure) created, for example, the screw surface of the rotating shaft and preventing the reduction of pressure in the working chamber of the main unit [1].

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Dry gas seals (DGS) are applied relatively recently. Nowadays more than 80% of the fleet of centrifugal compressors (turboexpanders) are equipped with such devices and the final version of standard for their device API 614, "Lubrication, Shaft-Sealing, and Control Oil Systems and Auxiliaries for Petroleum, Chemical, and Gas Industry Services" was developed by the American Petroleum Institute only in 1999, Improving the DGS is a relevant scientific, engineering and technological task.

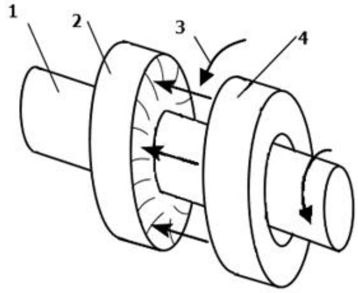

Fig.1. Sealing step of DGS.

The main working part of the sealing step is a pair of sealing (fig.1), one part of which is fixed bearing (4) made of high quality graphite with anti-friction treatment and “tighten” in the axial direction with the spring to the bearing (2) and the second movable foot bearing (2) is made of carbide material and mounted on a shaft 1 of the compressor.

Foot bearing 2 in the axial direction is fixed, at its working surface is made of spiral grooves, so when the shaft rotates, the process gas is captured by the grooves 3 and is blown into the gap between the foot bearings 2 and 4. This creates the required dynamic gas compression that reduces to an acceptable level, the output of the working gas through the gap of the bearing is between the shaft and the inner cylindrical surface of the bearing 4.

The obvious disadvantage of this device is the dependence of the characteristics of the seal on the speed of rotation of the shaft 1. During variable speed or reverse rotation this device will enter the mode of dry friction, and will have bad indicators of reliability. To eliminate this disadvantage and to reduce the output of the working gas it is suggested making compound bearing 4 so that on part of her internal cylindrical surface is incarnated waveformated motion (running wave) [2]. The direction of movement of running wave should be in the direction of the working gas with amplitudes commensurate with the size of the gap and the gradients are so large that the added pieces could have fit at least one or two wavelengths.

This seals is most effective while working with liquids, and the dynamic range of the generated backpressures is determined by the amplitude and frequency of the running wave. At very high speeds of wave motion in the fluid the cavitation can start, it reduces its operational capabilities, so in this report the characteristics of the compression with two-phase fluid are considered.

2. The equation for the pressure distribution.

A standard unit is selected as the main research object, two oppositely directed symmetric running waves (fig. 2), in the plane of symmetry which is observed zero consumption of lubricant. The dimensionless gap function we write in the following way:

$$H_i = \frac{h_i}{h_0} = \left\{ \begin{array}{c} 1 + e\cos\theta + A_0 \cos 2\pi\left(\tau + r\right), \quad i = 1 \\ 1 + e\cos\theta + A_0 \cos 2\pi\left(\tau + r\right)\cos 2\pi\left(\dfrac{\nu}{\omega}\tau + \phi\right), \quad i = 2 \end{array} \right\}, \tag{1}$$

in which the notation means: $r = z / \lambda$ – dimensionless axial coordinate; $\beta = \overline{B} / \lambda$ is the dimensionless length of the layer; $E_o = \overline{E} / h_o$ – the dimensionless oscillation amplitude; h0 is the nominal gap in a coaxial position of the cylinder; ω, ν is the frequency of the moving wave and the normal clearance of high-frequency vibrations, respectively.

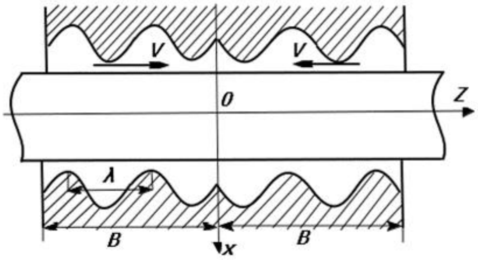

Fig. 2. Scheme of a gap.

The function H1 corresponds to a gap with a running wave, and H2 – the gap with the modulated running wave, $\tau = \nu t$.

Taking into consideration that that gas content in the liquid has the different nature – from the gap cavitation to the mechanical foaming, we describe the state of the environment with the help of universal dependencies of the following form: density of the medium $\rho = \rho_1 + \alpha_1(\rho_0 - \rho_1)$, where ρ0 and ρ1 – density of gas and liquid; the viscosity of the medium $\mu = \mu_1 + \alpha_1(\mu_0 - \mu_1)$, where μ0 and μ1 is the dynamic viscosity of the gas and liquid; the dependence between density and pressure $\rho = \kappa P^{\alpha 3}$ [3], where P is the pressure in the layer, α1, α2 α3 are the parameters of the void fraction close to each other and determined in each case according to the experiment. With this approach, the gas is evenly distributed by volume of the layer. Presented dependences uninterruptedly describe the state of the environment from an incompressible fluid (α =0) to pure gas (α =1).

We assume that the gap is thin enough for studying it only using small Reynolds numbers and flat flow pattern of liquid medium. We also accept the assumptions common to the theory of hydrodynamic lubrication of bearings [4], and, in particular, we neglect mass and inertia forces, the effect of the interaction between viscosity and compressibility. Then the equation for the pressure distribution, with taking into account some assumptions about the medium, is following:

$$\frac{\partial}{\partial r}\left[\left(\frac{H_i^{\frac{2\alpha-1}{\bar{\alpha}}}}{1+\alpha}\frac{\partial \Psi^{1+\alpha}}{\partial r} - \frac{1}{2\alpha-1}\Psi^{1+\alpha}\frac{\partial H_i^{\frac{2\alpha-1}{\bar{\alpha}}}}{\partial r}\right)\right] = \Lambda_\nu \frac{\partial \Psi^\alpha}{\partial \tau} \ , \tag{2}$$

where $\Lambda_\nu = 12\mu\nu R^2/(P_0 h_0^{\ 2})$, $\Psi^\alpha = P^\alpha H_i$ – required function of pressure [3]. Boundary conditions for the pressure can be obtained, at one end, as a condition of the free edge and at the geometric center, because of the complete symmetry must be equal to zero, local flow rate of the lubricant in the axial direction:

$$\Psi^\alpha = H_i(r=\beta,\tau) \ , \ \left(\frac{H_i^{\frac{2\alpha-1}{\bar{\alpha}}}}{1+\alpha}\frac{\partial \Psi^{1+\alpha}}{\partial r} - \frac{1}{2\alpha-1}\Psi^{1+\alpha}\frac{\partial H_i^{\frac{2\alpha-1}{\bar{\alpha}}}}{\partial r}\right)_{r=0} = 0 \ . \tag{3}$$

As initial conditions it is assumed:

$$\Psi\left(r,\tau=0\right) = 1 \ . \tag{4}$$

For the steady-state time-of course it is also true that the condition of periodicity in time

$$\Psi\left(r,\tau+1\right)=\Psi\left(r,\tau\right) \ . \tag{5}$$

The obtained mixed boundary value problem is nonlinear, which is typical for tasks of a thin layer with a compressible fluid. In the theory of gas lubrication function $\Psi = PH$ is used for the production primarily of private analytical solutions [3], which is associated with low sensitivity of this function to the change of the gap H. In the numerical solution this property is also useful to ensure the convergence of the task.

We use iterative scheme with linearization by Newton's method [5] for the solution of nonlinear mixed boundary value problem (2)…(5) and we build the iterative process using the following scheme

$$\dot{L}_{\Psi_0}(\Psi_{n+1}) = \dot{L}_{\Psi_0}(\Psi_n) - L(\Psi_n) \ , \tag{6}$$

where $\dot{L}_{\Psi_0}$ is the Frechet derivative from the space-time differential operator L , defined by equation (2); Ψ_0 – the initial approximation for the desired function of the pressure; Ψ_n and Ψ_{n+1} – n and n+1 the iterative approximation. As a result, the equation (2) corresponds to the following iterative equation:

$$\frac{\partial}{\partial r}\left\{ \frac{1}{\alpha}\left[H^{\frac{2-\alpha}{\alpha}} \frac{\partial \tilde{\Psi}_0^{\frac{1}{\alpha}} \tilde{\Psi}_{n+1}}{\partial r} - \frac{(1+\alpha)}{2\alpha-1}\tilde{\Psi}_0^{\frac{1}{\alpha}}\tilde{\Psi}_{n+1}\frac{\partial H^{\frac{2\alpha-1}{\alpha}}}{\partial r}\right] - \Lambda_v \frac{\partial}{\partial \tau}\tilde{\Psi}_{n+1}H^{1-\alpha} \right\} =$$
$$= \frac{\partial}{\partial r}\left\{ \left[\frac{1}{\alpha}\left(H^{\frac{2-\alpha}{\alpha}} \frac{\partial \tilde{\Psi}_0^{\frac{1}{\alpha}}\tilde{\Psi}_n}{\partial r} - \frac{(1+\alpha)}{2\alpha-1}\tilde{\Psi}_0^{\frac{1}{\alpha}}\tilde{\Psi}_n \frac{\partial H^{\frac{2\alpha-1}{\alpha}}}{\partial r}\right)\right] - \left(\frac{H^{\frac{2-\alpha}{\alpha}}}{1+\alpha}\frac{\partial \tilde{\Psi}_n^{\frac{1+\alpha}{\alpha}}}{\partial r} - \frac{(1+\alpha)}{2\alpha-1}\tilde{\Psi}_n^{\frac{1+\alpha}{\alpha}}\frac{\partial H^{\frac{2\alpha-1}{\alpha}}}{\partial r}\right)\right\} \tag{7}$$

where we introduced the notation $\tilde{\Psi} = \Psi^{\alpha}$.

For construction of the discrete equations the idea of balanced methods is used [6, 7]. The procedure to solve a similar problem was already described by us in [8] and will not be discussed here. As a result the dimensionless excess pressure in the layer is:

$$P(r,\tau) = \Psi / H_i^{\frac{1}{\alpha}} - 1 \tag{8}$$

and the average for the period of the running wave pressure is

$$\overline{P}(r) = \int_0^1 P(r,\tau)d\tau \cdot \tag{9}$$

To evaluate the delivery properties of gaps we use the value of the average pressure drop

$$\Delta\overline{P} = \overline{P}(0) - \overline{P}(\beta) \ . \tag{10}$$

3. The analysis of calculation results.

Some results of calculations are shown in fig. 3...5. In fig. 3 (a) the distribution along the axis of the cylinder instantaneous pressure in the gas layer is shown for values of the frequency parameter $\Lambda_\omega = 0,5$ and $\Lambda_\omega = 30$ with the dimensionless amplitude of E_0=0,35. As you can see, for large values of frequency parameter profiles plots of the pressure distribution again, mostly in the form of oscillations of the active surface. The compressibility of the medium leads to the asymmetry of the profiles of pressure. It is obvious that after the change of the gap in time is instantaneous local pressure, which determine the occurrence of excess pressure and medium. With the growth parameter Λ_ω reduces the width of the boundary zone, increases the pressure gradients in it and decreases the discharge pressure $\Delta\bar{P}$ of the medium in the inner region.

The distribution of the average for the period of the running wave pressures in the gas layer is presented in fig. 3 (b). For comparison the dependence for values of the frequency parameter is shown $\Lambda_\omega = 0,5$ and $\Lambda_\omega = 30$. From dependency of average over the period of pressures drop from the frequency parameter Λ_ω, which is shown in fig. 4 (a), we understand that their value reaches a maximum and then decreases with increasing Λ_ω, tending to some limiting value. This suggests that, unlike gaps, filled with incompressible liquid, where the conditions of continuity of the medium there is a continuous increase of pressure with increase Λ_ω, for a compressible medium the injection of the medium into the gap occurs only for small values of the frequency parameter.

The feature is that the maximum pressure drop is achieved at smaller Λ_ω and is a function of the frequency ratio ν/ω: ν /ω =10 the maximum is achieved at Λ_ω=2,5 and for ν/ω =5 – Λ_ω=3,5...4. This can be explained, if we imagine the function of gap H_2 in the following form:

$$H_2 = 1 + \frac{A_0}{2}\left\{\cos\frac{2\pi}{\omega}\left((\omega+\nu)\tau + r + \phi\right) + \cos\frac{2\pi}{\omega}\left((\nu-\omega)\tau - r + \phi\right)\right\} \quad (11)$$

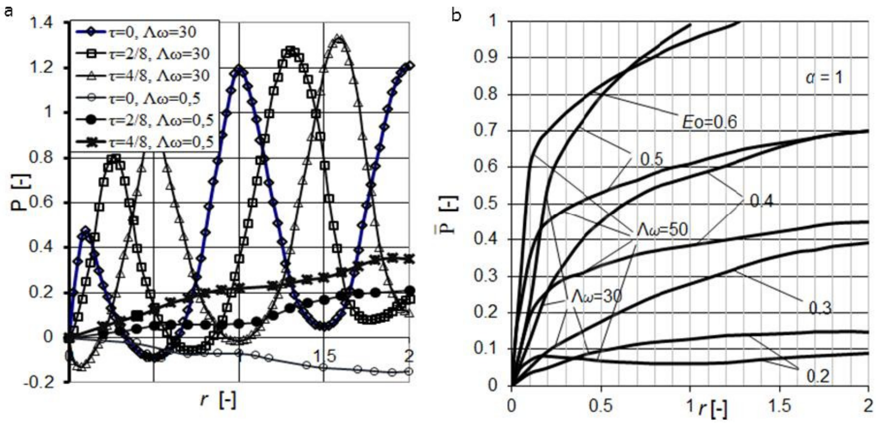

Fig. 3. The distribution of (a) instantaneous (*E*o=0,35; α=1) and (b) average for the period of the running wave pressures (α=1) in the layer.

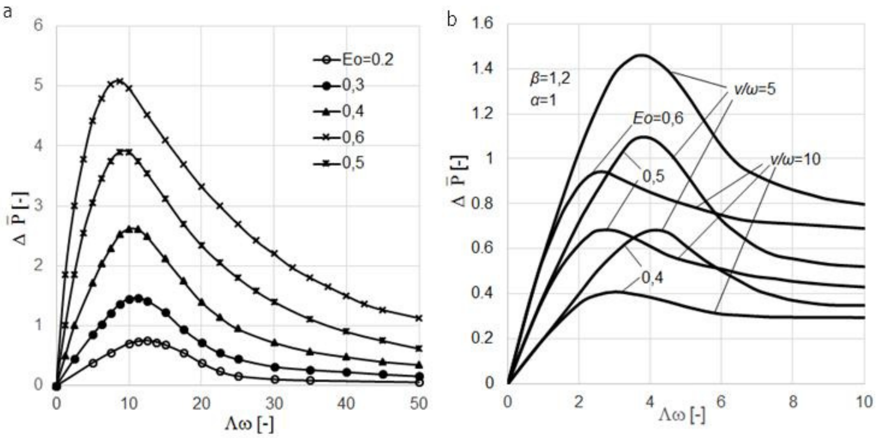

Fig. 4. Drop of pressure in the layer (a) with the moving wave (α=1; β=2) and (b) modulated moving wave (α=1; β=1,2).

We see that this “signal” contains a frequency spectrum with upper $\omega+\nu$ and lower $\nu-\omega$ side frequencies, so the upper side frequency and determines the effect of “blocking” layer, with maximum pressure drop observed for values approximately $(\omega+\nu)/\omega$ times smaller than for gaps with non-modulated running wave.

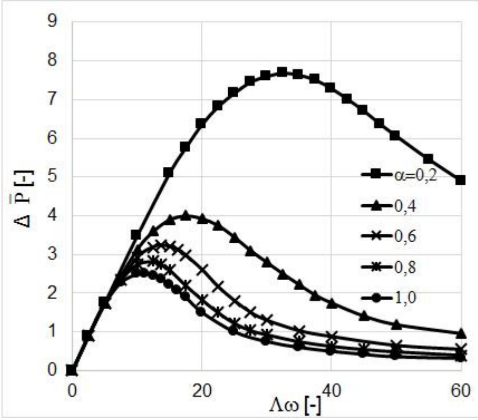

Fig. 5. Drop of pressure (backpressure) for different values of the parameter α is the void fraction (E_0=0,4; β=2).

The effect of the void fraction α by the amount of pressure drop can be seen in fig. 5 for the amplitude E_0=0,4 and β=2. In addition to increasing pressure in the region of small Λ_ω there is shear of maximum pressure towards larger values Λ_ω with decreasing void fraction α. The presence of an extremum is the result of a balance of forces is determined by the kinematics of the gap with a moving wave, and viscous friction forces impeding the fluid flow. Given that with increasing frequency of changes of the gap there is an increase in the forces of viscous friction [3], we get that for some sufficiently large values of frequency parameter Λ_ω for compressible medium completely stops, the injection of the medium into the gap is terminated and the action of the moving wave is reduced only to the known effect of compression of a compressible medium [3].

4. Conclusion.

It was obtained that the most efficient use of hydrodynamic seals with a running wave occurs while using incompressible fluids, where fluctuations generated back pressures can be, theoretically, infinitely large. A limiting factor may be cavitation, which will lead not only to reduction of the effective viscosity and density of the technological environment, but also to the appearance of compressibility. As a result, the presence of liquid, for example, 20 % of gas will cause reduction generated by the layer of pressure up to 8·105 [Pa] even at the optimum speed of a moving wave with amplitudes of 40 % of the gap.

However, in practice, with proper selection and preparation of technological liquid to expect it is much less gas and generated a layer of the pressure drop will depend on already on operational characteristics of the vibroengine.

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